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PERFORMANCE OF A COMBINED SOLAR DESICCANT FILTRATION AIR-CONDITIONING SYSTEM

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ABSTRACT

A solar desiccant filtration system and conventional system were compared by simulating their performance over a cooling season using TRNSYS and Typical Meteorological Year (TMY) weather data for a range of system parameters. The desiccant dehumidifier was modeled as a heat and mass exchanger with moisture and temperature effectivenesses based on the equilibrium intersection point. The activated carbon filter was modeled as a reduction of required ventilation flow rate to maintain acceptable contaminant levels in the space. Several locations were investigated. The desiccant isotherm shape was varied from a moderate Brunauer Type I to silica gel (linear) and the solar collector area was varied from 0 to 4,000 ft^2 to see its effect on the savings per season. The isotherm shape and location had little effect on the resulting savings. The savings were between \$2,800 and \$4,800 per season depending on the amount of collector area.

1. INTRODUCTION

Desiccant air-conditioning systems have been proposed as alternatives for conventional air-conditioning systems. These systems reduce the peak electric demand and electrical energy use. The reductions in electricity costs may be offset by increased natural gas consumption. Heat from a solar collector array with further reduce costs. However, the first cost of these systems is large compared to alternative conventional vapor compression systems, and their energy savings do not currently provide sufficient financial incentive. Filtration systems have been proposed as an alternative way to meet indoor air quality (IAQ) standards without excessive outdoor air flows. Combined desiccant and filtration systems can provide air-conditioning and meet IAQ standards at reduced electric demand and energy and improve the economics of solar-desiccant for cooling systems.

2. CYCLE DESCRIPTION

The desiccant system evaluated in this study incorporates a solar regenerated rotating desiccant dehumidifier operating in recirculation mode together with fixed bed filters to reduce the ventilation flow rate. The filtering system consists of an activated carbon filter in the supply air stream that reduces the volatile organic compounds (VOCs) entering the space. The filter is regenerated by the heat from the solar collector. A gas furnace is used when the flat plate solar collector cannot deliver the heat necessary to regenerate the desiccant. Fig. 1 shows the schematic of the solar desiccant filtration system.

The conventional system is a vapor compression air-conditioner with a coefficient of performance (COP) of 3.0 at ARI conditions.

The internal space was assumed to be a 10,000 ft^2 commercial office building with an occupancy of 100 persons. The total volume flow rate of air delivered to the space in both systems is 12,000 cfm. The reactivation flow rate in the solar desiccant filtration system is 8,000 cfm. An internal VOC generation of 5 mg/m^3 was assumed for the internal space. This is a representative value for commercial buildings, and a ventilation flow rate of 30 cfm/person would be required to provide sufficient indoor air quality (1, 2). The desiccant filtration system allows a reduction in outdoor ventilation flow to 15 cfm/person. Both systems are controlled to meet both the sensible and latent loads on the space.

3. COMPONENT MODELING

The conventional and desiccant systems were modeled using TRNSYS (3), a transient simulation program. The cooling season was assumed to be from May 1 to September 30 and was simulated using TMY weather data. The parameters of the systems are given in Tables 1 and 2.

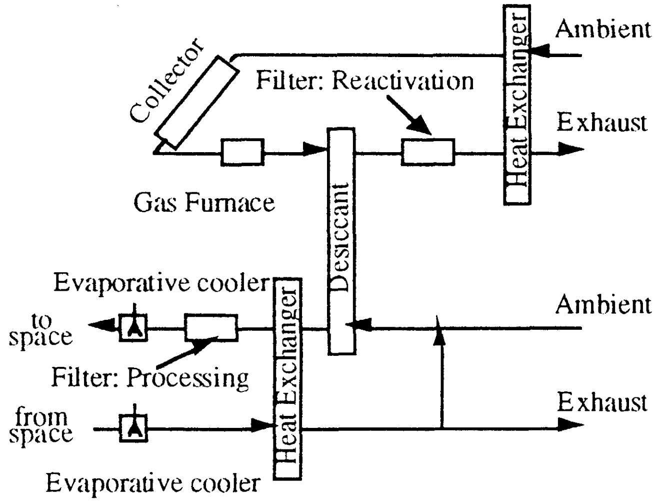

Fig. 1 Solar desiccant filtration system schematic.

3.1. Building Load

The building load was modeled by assuming that the load was made up of three components: 1) sensible and latent loads from the occupants of 250 and 175 Btu/h-person respectively, 2) sensible and latent loads from equipment of 3 W/ft^2 and 17,500 Btu/h respectively, and 3) sensible load from solar radiation. The solar radiation load was assumed to come only through the roof and was calculated using the sol-air temperature method (4). The roof conductance was assumed to be 0.1 Btu/h-ft^2-°F, the roof absorptivity was assumed to be 0.9, and the heat transfer coefficient on the roof was assumed to be 4 Btu/h-ft^2-°F. The resulting building load for Washington D.C was 280 MBtu/season with a maximum load of 20 tons and a sensible heat ratio of 0.8.

3.2. Air Conditioner Model

The air conditioner was modeled using a TRNSYS as an air cooled package unit. The rated capacity, COP, bypass fraction and volume flow rate at ARI conditions are input. The part load performance is determined by curve fits from DOE-2 models (5). The air-conditioner was controlled to meet both the sensible and latent load on the space.

3.3. Desiccant Model

The desiccant model used an effectiveness approach (6). The effectiveness of the desiccant dehumidification process was based on the equilibrium intersection point, and the reactivation outlet condition was determined from mass and energy balances on the dehumidifier. The humidity ratio effectiveness was assumed to be constant and the temperature effectiveness was assumed to be unity. The effectiveness are given as follows (6):

$$\varepsilon_w = \frac{w_{in} - w_{out}}{w_{in} - w_{int}} \quad (1)$$

$$\varepsilon_t = \frac{t_{in} - t_{out}}{t_{in} - t_{int}} \quad (2)$$

where the subscript int refers to the intersection point of the desiccant dehumidifier.

The nominal isotherm shape was for silica gel (7). A desiccant with a moderate Brunauer Type I isotherm shape was also simulated. The dehumidifier reactivation temperature was controlled so the dehumidifier exactly met the load on the space. For Washington D.C. the reactivation temperature was in the range of 110-170°F.

3.4. Solar Collector

The solar collectors were flat plate air heating units tilted at the latitude of the site. The collectors were modeled with constant $F_r(\tau\alpha)$ and F_rU_l of 0.66 and 1.06 respectively. A store for solar energy is not needed because air-conditioning is required only during day times.

3.5. Carbon Filter

The carbon filters were sized to adsorb the pollutants generated on a daily basis. The filters are reactivated using the exhaust from the rotary desiccant dehumidifier. For Washington D.C., the reactivation temperature was in the range of 90-120°F. The process temperature was about 70°F.

3.6. Heat Exchangers and Evaporative Coolers

The heat exchangers and evaporative coolers were modeled as constant effectiveness units. The effectiveness for the evaporative cooler is based on the dew point temperature.

3.7. Fans

The fan power is estimated by assuming a fan and motor efficiency and estimation of pressure drops through the system components. The condenser fan on the conventional system was assumed to be 10% of the compressor power.

4. SYSTEM PARAMETERS

The physical parameters used in the simulation of the conventional and the solar desiccant filtration system are shown in Table 1 and Table 2 respectively. The economic parameters used to determine the cost of operation per season are $0.05/kWh of electricity energy change, $15/peak kW per month demand charge, and $5/MBtu of natural gas.

Table 1 CONVENTIONAL SYSTEM PARAMETERS FOR WASHINGTON D.C.

Air conditioner	
COP	3.0
Capacity, Btu/h	550,000
Volume flow rate, cfm	12,000
Pressure Drop, % of compressor power	10
Rest of system	
Pressure Drop, inches of water	1.7

5. RESULTS

The results for the simulation are detailed in the following sections. The building is occupied only during daytime hours, and the air-conditioning is required only during that period. The system control was to first use solar to regenerate the desiccant, and then the gas furnace. The carbon filter is regenerated using the air discharge from the desiccant dehumidifier. The amount of solar collector area, isotherm shape and location were varied.

5.1 Solar Collector Area

The purchased gas for Washington D.C. as a function of the solar collector area is shown in Fig. 2. The amount of gas purchased decreases linearly with collector area for low collector areas, but as the area increases over 3,000 ft^2 the added benefit of adding collector area is decreased. This decrease is because there are times where the added collector area does not affect the amount of gas needed.

The savings per season were computed using the economic parameters listed earlier. The reduction in peak electric demand was about 49 kW per month and the reduction in electric energy was 34.5 MWh per cooling season in Washington D.C. The savings per season as a function of collector area are shown in Fig 3.

Table 2. DESICCANT SYSTEM PARAMETERS.

Desiccant Dehumidifier	
Effectiveness[1]	0.75
Pressure Drop Processing[2]	1.1
Reactivating	2.4
Carbon Filter	
Fraction[3]	0.5
Pressure Drop	0.2
Heat Exchangers	
NTU	3.0
Pressure Drop	0.5
Gas Furnace	
Efficiency	0.90
Pressure Drop	0.6
Solar Collector	
$F_r(\tau\alpha)$	0.66
F_rU_l	1.06
Pressure Drop	0.75
Evaporative Coolers	
Effectiveness	0.85
Pressure Drop	0.60
Fan + Motor	
Efficiency	0.70

Notes:
1. Humidity ratio effectiveness
2. All pressure drops in inches of water.
3. Fraction refers to the fraction of the nominal ventilation flow rate.

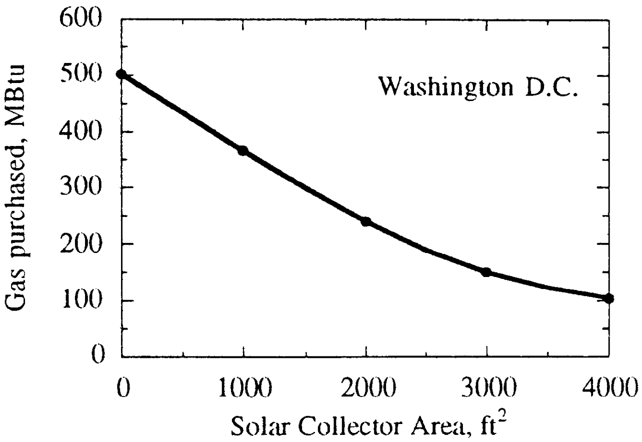

Fig. 2 Gas purchased as function of collector area.

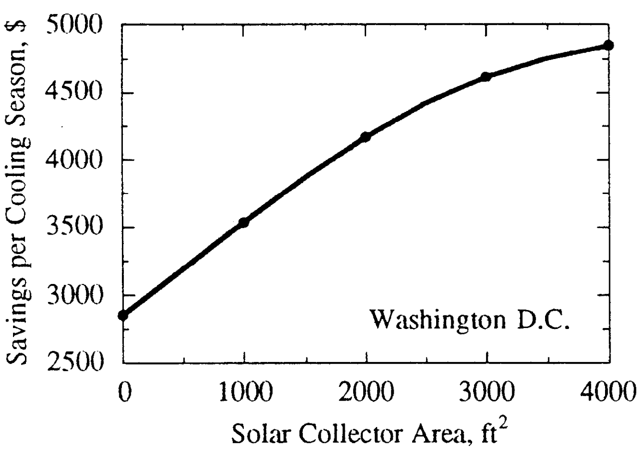

Fig. 3 Savings per cooling season as function of solar collector area.

5.2 <u>Location of Building</u>

The systems were simulated at four locations to determine the effect of the site for the solar desiccant filtration system on the seasonal savings. The solar collector area for all the sites was 2,000 ft^2. The sites chosen were Washington D.C.; St. Louis, MO; Chicago, IL; and Miami, FL. Fig. 4 shows the difference in average electric demand reduction, seasonal electric energy reduction and seasonal purchase gas for the locations. There are significant reductions in peak demand. The COP of the conventional vapor compression system is lowest during the hottest weather, therefore, the demand savings are greatest under conditions for which the utility capacity is lowest. Fig. 5 shows the savings per season. The savings per season at different locations varied by less that 10%. The increased savings from the larger reduction of electrical energy consumption in Miami was offset by the cost of increased gas consumption.

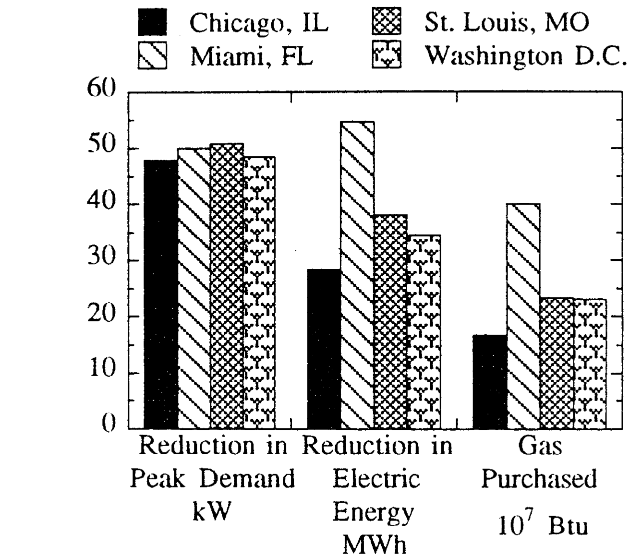

Fig. 4 Reduction of electric peak demand and energy and gas purchased for several locations with 2,000 ft^2 of solar collector.

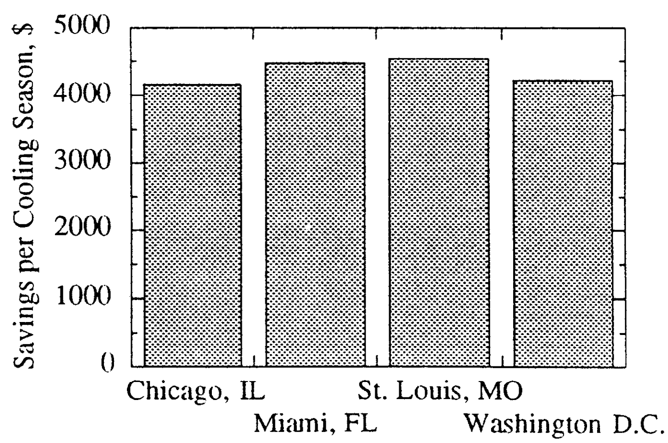

Fig. 5 Savings per season for several locations with 2,000 ft^2 of collector.

5.3 <u>Desiccant Isotherm Shape</u>

The desiccant isotherm shape was varied between a linear isotherm and a moderate Brunauer Type I. The was a small improvement with the Type I isotherm, but both dehumidifier types offer significant savings over conventional air-conditioning systems.

6 <u>CONCLUSIONS</u>

The solar desiccant filtration system significantly reduces peak electric demand and energy. For Washington D.C., the reduction of peak electric demand was about 49 kW per month and the reduction of electric energy was 34.5 MWh.

The additional costs incurred are for gas consumption for the furnace. The result is savings of \$2,800 to \$4,800 per cooling season depending on the amount of solar collector area. The savings per season as a function of building location were all in the range of \$4,100 to \$4,500 for 2,000 ft^2 of solar collector area. The change in desiccant isotherm shape from linear to moderate Brunauer Type I had an insignificant effect on the seasonal savings. These systems appear to be economically viable because the filtration system significantly reduces the outdoor ventilation requirement.

7. NOMENCLATURE

Roman variables

t Temperature

w Humidity ratio

Greek variables

ε_t Temperature effectiveness for desiccant dehumidifier

ε_w Humidity ratio effectiveness for desiccant dehumidifier

Subscripts

in inlet

out outlet

int intersection point

8. REFERENCES

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